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RESEARCH MEMORANDUM

EXPERIMENTAL INVESTIGATION OF STAGE PERFORMANCE OF J71 THREE-STAGE TURBINE

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NATIONAL ADVISORY COMMITTEE
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WASHINGTON

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SUMMARY

An experimental investigation of the stage performance of the J71 three-stage turbine at design speed and pressure ratio was conducted as part of a program involving research on the design and operational characteristics of high-work-output multistage axial-flow turbines.

The investigation indicated that the over-all work output of the turbine based on temperature measurements at design speed and pressure ratio was 95 percent of the design value, with the greatest loss in work output and efficiency occurring in the first stage.

Choking in the second-stage stator over most of the pressure ratios investigated prevented the attainment of design first-stage work output and also prevented any increase in work output of the first stage with increasing over-all pressure ratio at design speed. High losses in the first stage were a contributing factor to this choking condition.

INTRODUCTION

As part of a general research study of high-work-output, low-speed, multistage, axial-flow turbines at the NACA Lewis laboratory, the stage performance of the J71 three-stage turbine was experimentally investigated. A previous investigation of over-all performance (ref. 1) showed that the efficiency at design speed and pressure ratio was considerably lower than expected, with the result that only 95 percent of design work was obtained at these conditions. It was desirable, therefore, to determine which blade row or rows were responsible for this loss in efficiency. Consequently, interstage instrumentation was utilized to obtain limited measurements between blade rows at design speed and pressure ratio in order to study this problem. Qualitative radial variations of individual stage work output and adiabatic efficiency were obtained from these measurements. The measurements consisted, in general, of inner- and outer-wall static pressures and of temperature, stagnation pressure, and flow

angle at various radial locations. Because of the complex nature of the instrumentation required in a survey investigation of a multistage turbine, the measurements were limited, in general, to one or two circumferential positions. Although it is recognized that large circumferential variations in flow (refs. 2 and 3) downstream of a blade row exist because of three-dimensional flow effects such as secondary flows, finite number of blades, and so forth, it was felt that the simplified instrumentation used would probably reveal any major malfunctioning of a particular blade row. Thus the measurements are interpreted as indicating trends of relative performance between the turbine stages in a qualitative manner.

The stage performance is presented in terms of an individual stage efficiency and a work parameter corresponding to an equivalent stage temperature drop. Regions of stage performance deficiencies are indicated and discussed.

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SYMBOLS

The following symbols are used in the report:

A	annular area, sq ft
g	gravitational acceleration, 32.174 ft/sec^2
h	enthalpy, Btu/lb
J	mechanical equivalent of heat, 778.2 ft-lb/Btu
p	pressure, lb/sq ft
T	temperature, $^\circ$R
V	absolute gas velocity, ft/sec
γ	ratio of specific heats
η	adiabatic efficiency
ρ	gas density, lb/cu ft

Subscripts:

1,2,3,4,5,6,7	measuring stations (see fig. 1)
av	mass-averaged value

e	exit from a blade row
i	inlet to a blade row
u	tangential
x	axial

Superscript:

'	stagnation or total state

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APPARATUS, METHODS, AND PROCEDURE

The experimental test installation, the method of power absorption, and the apparatus are as described in reference 1 with the exception of the survey apparatus. This consisted of movable, unshielded, claw-type, stagnation-pressure probes with provision for angle measurement. These were used to measure radial variations in flow angle and stagnation pressure through remote-controlled actuators located at each measuring station as shown in figure 1. The pressure-angle probe employed was of the type depicted in figure 2. Static pressures were determined by assuming a linear radial variation between the values measured from wall static taps on the inner and outer shrouds behind each blade row.

The temperatures were measured by fixed calibrated spike-type thermocouple rakes consisting of five thermocouples at the area centers of five equal annular areas located at different circumferential positions as shown in figure 1.

The survey data were obtained by operating the turbine at the equivalent design speed of 3028 rpm and equivalent design pressure ratio of 3.50. Reference 1 shows that equivalent design work was obtained at a higher pressure ratio, but the value of 3.50 used herein was in the range of high efficiency. Since it is also the design pressure ratio of the turbine, it was considered the proper value to use in this investigation. The inlet stagnation pressure was 40.5 inches of mercury absolute and the inlet temperature was 700° R. Radial surveys were taken at seven radial positions corresponding to various percentages of annular area at survey stations located as shown in figure 1.

Stage work parameter. - The work output of a stage is expressed as an equivalent stage temperature drop $(T_i' - T_e')/T_i'$ where the denominator is the stage inlet absolute stagnation temperature which corresponds to that at the exit of the preceding stage.

Stage and over-all efficiencies. - The turbine stage and over-all adiabatic efficiencies at various radial positions were calculated by the equation

$$\eta_{i-e} = \frac{1 - \frac{T'_e}{T'_i}}{1 - \left(\frac{p'_e}{p'_i}\right)^{\frac{\gamma-1}{\gamma}}} \tag{1}$$

in which the pressure and temperature measured at radial positions corresponding to the same percentage annular area at the inlet and exit of a stage are used.

Mass-averaged values. - The mass-averaged values of the various parameters used in the investigation are obtained from the equation

$$\left(\frac{T'_i - T'_e}{T'_i}\right)_{av} = \frac{\int_0^A \frac{T'_i - T'_e}{T'_i} \rho V_x dA}{\int_0^A \rho V_x dA} \tag{2}$$

Here the parameter is $(T'_i - T'_e)/T'_i$. The same method is used to calculate mass-averaged values of efficiency. For example, the mass-averaged efficiency is obtained by calculating the values of efficiency and specific mass flow at a given percentage of annular area from the measured pressure, temperature, and flow angle at the particular radial position. The product of the efficiency and specific mass flow is then numerically integrated with respect to annular area, and the result is divided by integrated specific mass flow to give the mass-averaged efficiency.

Exit whirl. - The exit tangential velocity is calculated by applying the experimentally determined pressures, temperature, and flow angle at the turbine exit to the energy equation for one-dimensional compressible flow.

RESULTS AND DISCUSSION

The experimental stage performance of the J71 turbine is presented in terms of a work parameter involving stagnation temperature drop and an adiabatic efficiency plotted against percentage of annular area, hereinafter referred to as "radial coordinate."

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Turbine work parameter. - The over-all work parameter and that of each stage are shown in figure 3. The mass-averaged values of the work parameter are compared with the respective design values. The design value of the over-all and stage work parameters is obtained by using the design equivalent over-all work output and the design percentage work for each stage. The design turbine work split was 38.5 percent, 33.0 percent, and 28.5 percent in the first, second, and third stages, respectively (ref. 1). From these values each stage temperature drop is determined and then divided by the corresponding stage inlet temperature in a stage-by-stage calculation to give the particular stage work parameter.

The variation of the over-all work parameter with the radial coordinate is shown in figure 3(a). Dotted lines on this and subsequent figures indicate data in the blade end regions which are of questionable accuracy. The mass-averaged value of the over-all work parameter is 0.247, which is approximately 95 percent of the design value. This agrees with the over-all work output based on torque measurements, which was also 95 percent of the design value.

The radial variation of work output in the first stage is shown in figure 3(b). The mass-averaged value of 0.085 is approximately 85 percent of the design value. The variation of work parameter with radial coordinate in the second and third stages is shown in figures 3(c) and (d), respectively. The mass-averaged value of the second-stage work parameter of 0.092 is approximately 97 percent of the design value, while that of the third stage is 0.095, which is approximately 4 percent greater than the design value. Thus the first stage, which produces only 85 percent of design stage work, appears to be the source of the major work deficiency in the turbine. In figure 3(b) it is observed that the radial decrease in work parameter toward the tip for the first stage is more pronounced than for either the second or third stage. This gradient near the tip is also much greater than the most extreme of those at any particular circumferential position observed in reference 2. This indicates that the first-stage work deficiency is caused by phenomena other than those which can be shown by detailed circumferential measurement. Thus, additional surveys at other circumferential positions would probably not alter the results significantly.

In general, the variations in the stage work parameter indicate that the second and third stages produce approximately design work, while the first stage is mainly responsible for the over-all work deficiency.

Turbine adiabatic efficiency. - The radial variation of over-all and individual stage adiabatic efficiency as calculated by equation (1) is shown in figure 4. The over-all efficiency variation with radial coordinate is represented in figure 4(a). The mass-averaged value of the over-all efficiency is 0.83.

The radial variation of efficiency in the first stage is shown in figures 4(b) and (c). The mass-averaged value of first-stage efficiency is approximately 0.74.

Because flow measurements at the inlet to the first stage were taken at a station which is a substantial distance upstream of the inlet-stator leading edge (fig. 1), it is evident that part of the loss in the first stage occurs in the inlet transition section between measuring station 1 and the first-stator leading edge. In order to determine the effect of possible inlet section losses on first-stage efficiency, the efficiency of the first-stage rotor was calculated independent of the rest of the stage and is shown in figure 4(c). The radial variation of the first-stage rotor efficiency is very similar to that of the entire stage and indicates approximately a 3-percent loss in stage efficiency through the inlet section and stator, since the rotor mass-averaged efficiency is approximately 0.77 as compared with 0.74 for the stage.

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The second- and third-stage efficiency variations with radial coordinate are shown in figures 4(d) and (e), respectively. The second-stage mass-averaged efficiency is approximately 0.81, while that of the third stage is approximately 0.89. In figure 4(e) the third-stage efficiency reaches a maximum of slightly greater than 100 percent at the 35-percent radial coordinate and decreases to approximately 65 percent at the blade ends. The value greater than 100 percent is probably the result of radial displacement of streamlines other than that assumed through the stage. This displacement causes the inlet and exit pressures and temperatures measured at a given percentage annular area to be incorrect when used in equation (1), since it is valid only along a streamline.

In general, it is noted that the calculated over-all efficiency (0.83) agrees with the value of 0.83 based on torque measurements reported in reference 1. The radial and stage variations in efficiency also indicated that the first stage was the major source of poor performance in the turbine.

Exit whirl loss. - Normally the kinetic energy $V_{u,7}^2/2gJ$ at the turbine exit (station 7, fig. 1) is not available for producing useful thrust; thus the turbine is penalized for the amount of this energy when the efficiency is calculated. Because this energy is then considered lost, the possible effect of the energy (expressed as a fraction of the total work output) on the magnitude of the efficiency is shown in figure 5. The mass-averaged value of the energy contained in the tangential velocity component is approximately 0.3 percent of the total turbine enthalpy drop and is thus a negligible quantity for operation at design speed and pressure ratio.

Turbine choking point. - If choked flow occurs in a blade row of a multistage turbine at a given total-to-static pressure ratio, an increase

in the pressure ratio, that is, a decrease in static pressure at the blade exit, will produce no static-pressure change upstream of the choked blade row. Thus, if a multistage turbine chokes first in an upstream blade row, another blade row or subsequent blade rows downstream will eventually choke with continual increase of the over-all turbine pressure ratio. Likewise, if a downstream blade row chokes first, conditions at the upstream blade rows remain unchanged with further increase of over-all turbine pressure ratio.

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In order to illustrate choking conditions at design speed in the J71 three-stage turbine, the ratio of inlet total pressure to static pressure at the tip at a given station is plotted against over-all turbine pressure ratio in figure 6 for each measuring station. These pressure data were recorded during the performance investigation reported in reference 1. If, for a particular pressure ratio, the curve for a given station has a definite positive slope, while the curve for the preceding station has zero slope, it is evident that the blade row between those stations is choking. Thus it can be seen in figure 6 that the second stator (between stations 3 and 4) is choked over approximately the entire range of over-all pressure ratios. This is partially caused by the losses in the first stage indicated by the surveys. These losses produce a low gas density and thus high velocity at the second-stator inlet which results in choking before the design work output is obtained. Thus further increase in over-all turbine pressure ratio could not increase the deficient work output of the first stage. Consequently, design over-all turbine work was obtained only by extracting greater-than-design stage work from the third stage by operating the turbine at an over-all pressure ratio approximately 6-percent higher than the design value. The third stage thus operated at a condition approaching that of limiting loading in order to obtain design work, as is shown in figure 7 of reference 1.

It can also be seen in figure 6 that the curve for station 2 indicates no choking in the first-stage stator because, although the curve has zero slope, the pressure ratio p_1'/p_2 is approximately 1.40, about 25 percent less than the theoretical nozzle choking value of 1.89.

CONCLUDING REMARKS

An investigation of the stage performance of the J71 three-stage turbine at the design equivalent conditions of 3028 rpm and 3.50 over-all pressure ratio indicated that:

1. The over-all work output based on temperature measurements was 95 percent of the design value and the over-all efficiency was 0.83.

2. The first stage exhibited the greatest deficiency in performance with an efficiency of only 0.74 and a work output only 85 percent of the design value.

3. Design work output for the first stage was not obtained because the second-stage stator was choked over most of the range of pressure ratios investigated. This choking condition also prevented any increase in first-stage work output with increase in over-all turbine pressure ratio. High losses in the first-stage rotor were a contributing factor to this choking condition.

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National Advisory Committee for Aeronautics
Cleveland, Ohio, September 9, 1954

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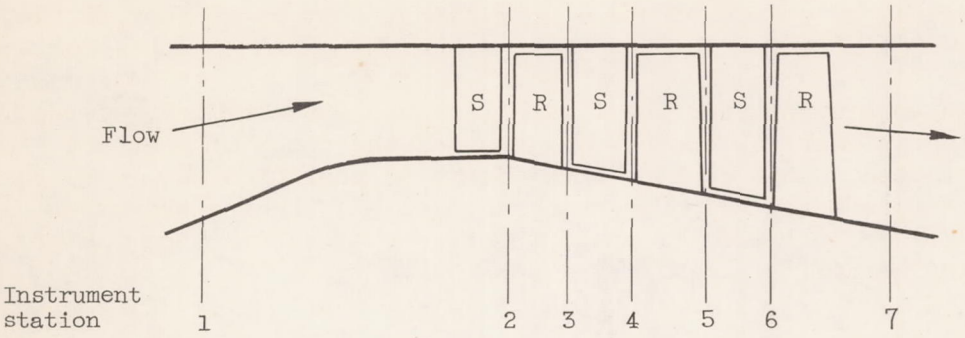
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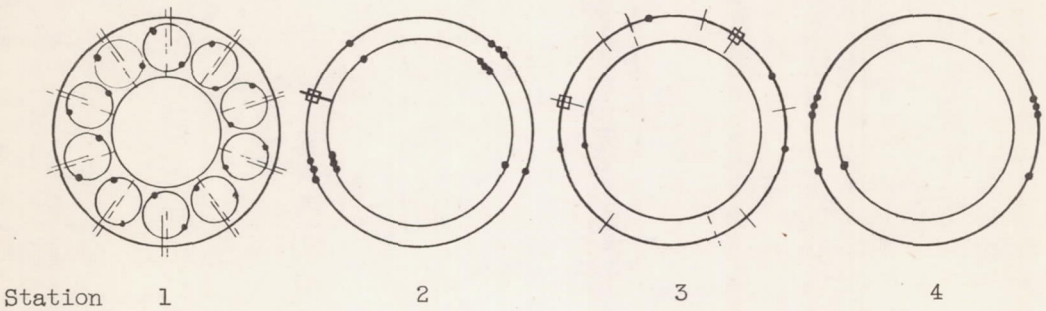
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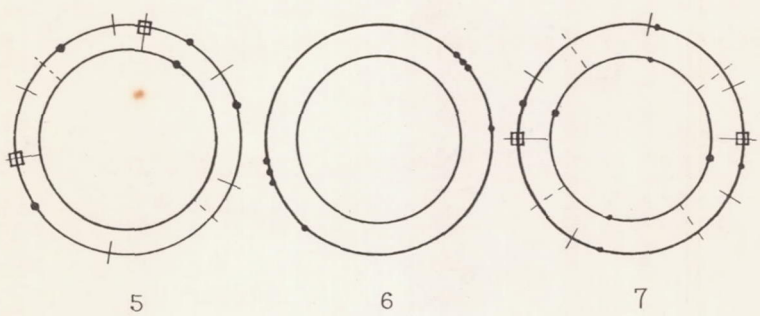
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CD-3807

Movable probe (to measure angle and stagnation pressure)
Stagnation pressure
Static pressure
Stagnation temperature
S Stator
R Rotor

Figure 1. - Schematic diagram of J71 three-stage turbine flow passage in a radial-axial plane showing instrument stations and location of instruments at each station.

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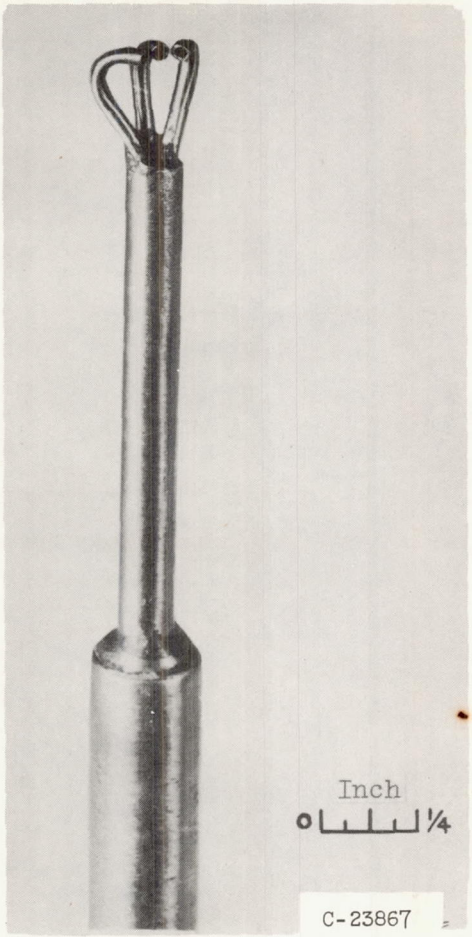
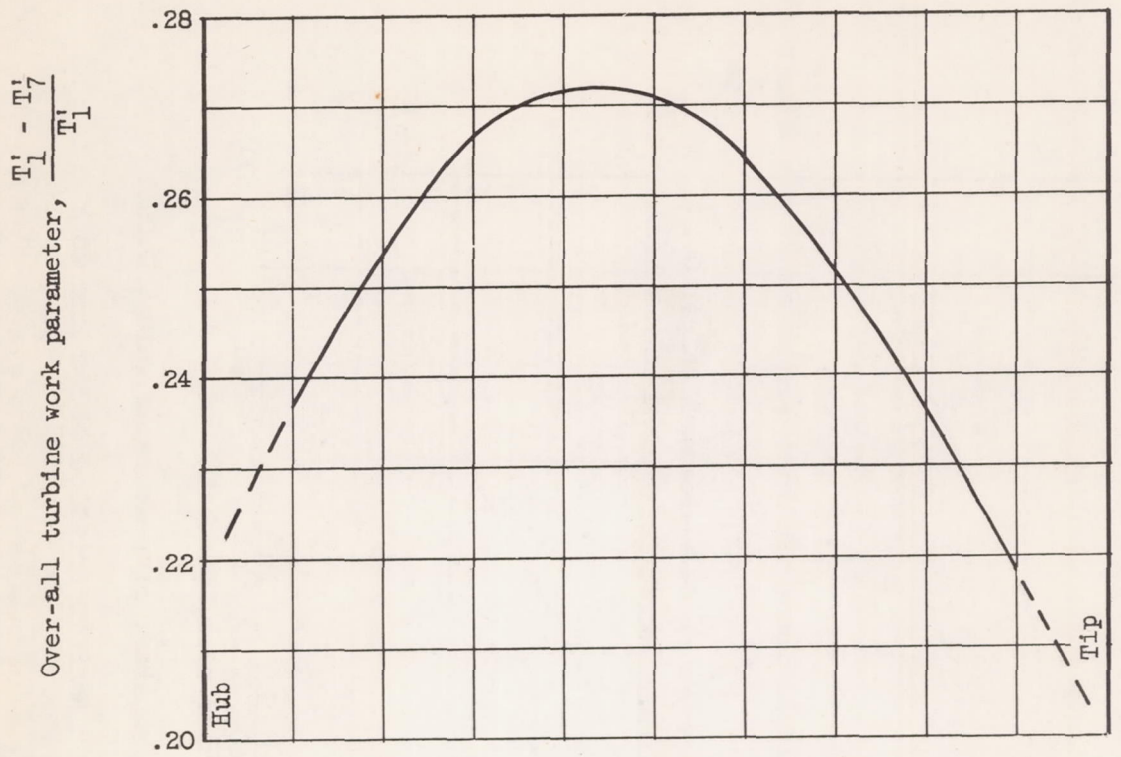
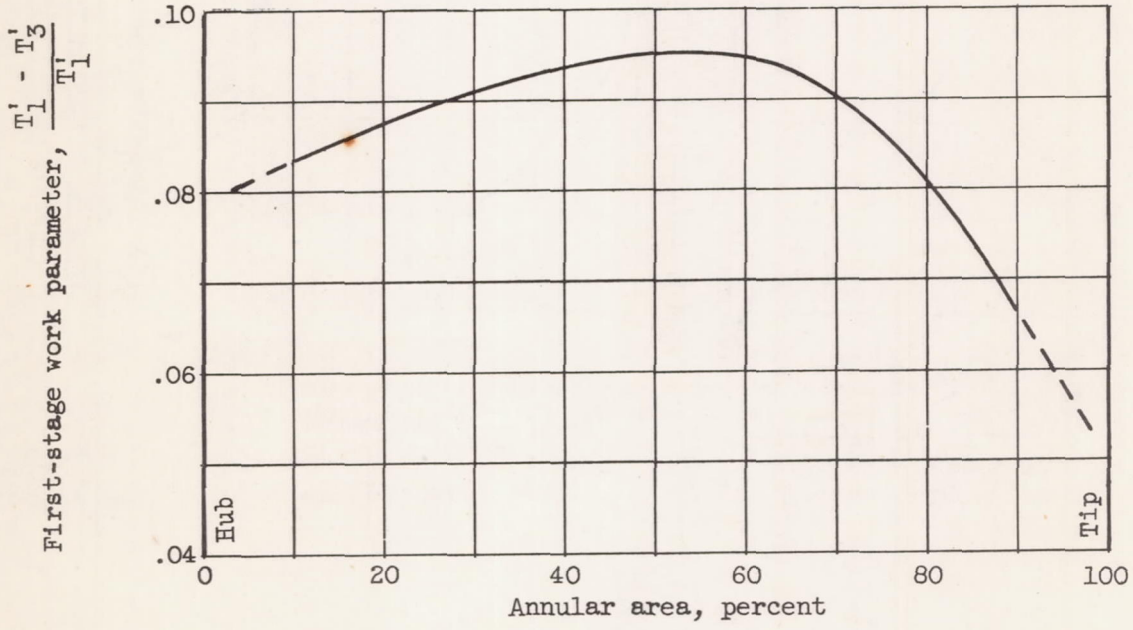


Figure 2. - Stagnation-pressure claw probe used to measure radial variations in stagnation pressure and angle.

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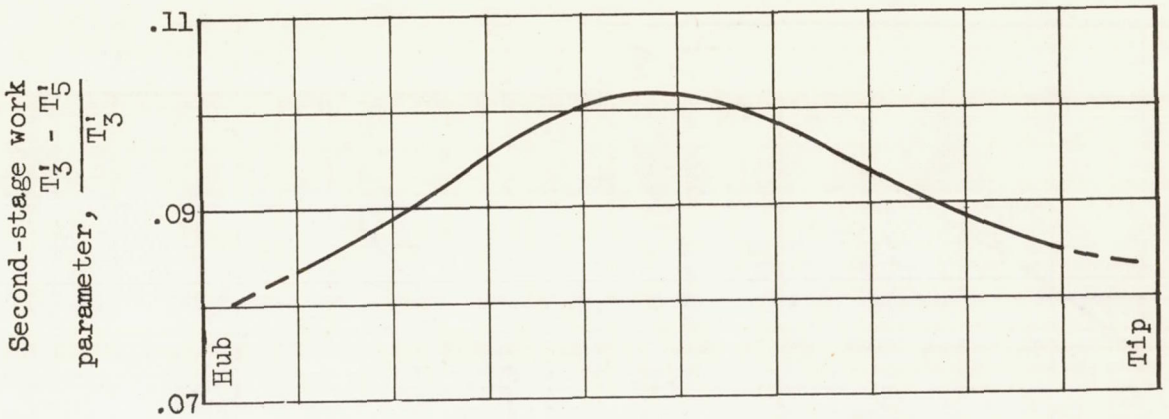


(a) Over-all work parameter; mass-averaged value, 0.247.

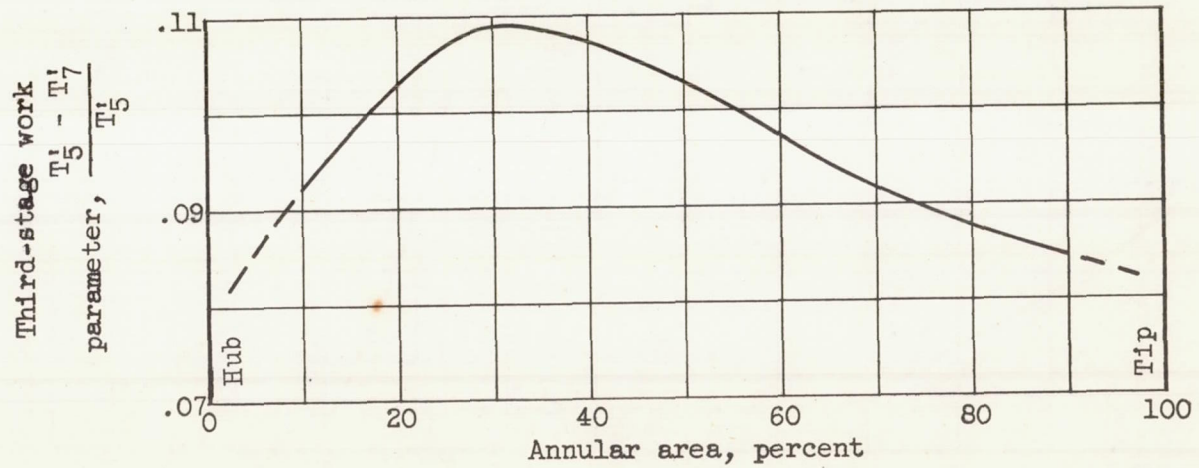


(b) First-stage work parameter; mass-averaged value, 0.085.

Figure 3. - Radial variation of turbine work parameter.



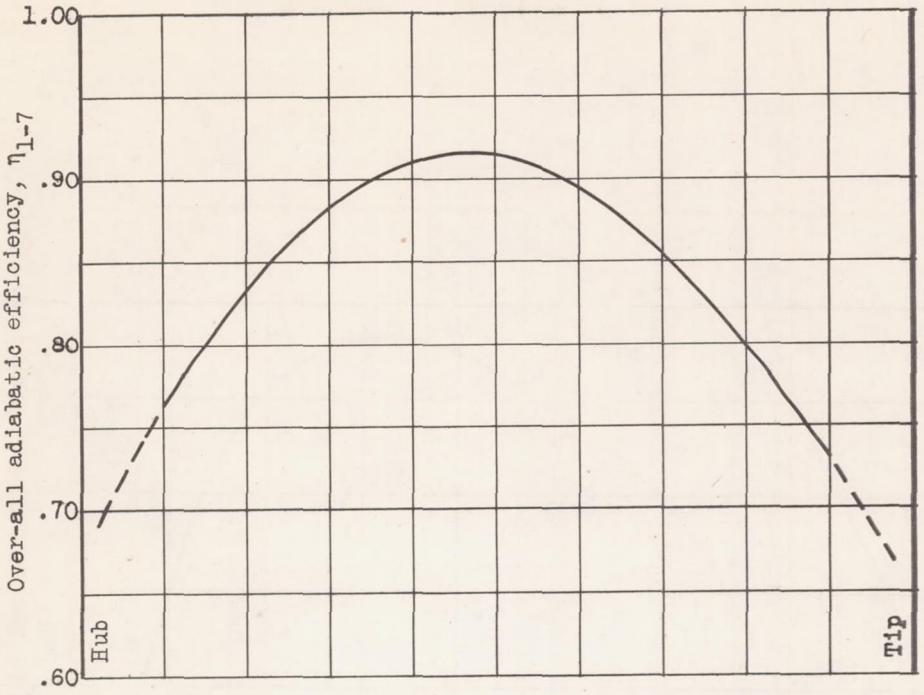
(c) Second-stage work parameter; mass-averaged value, 0.092.



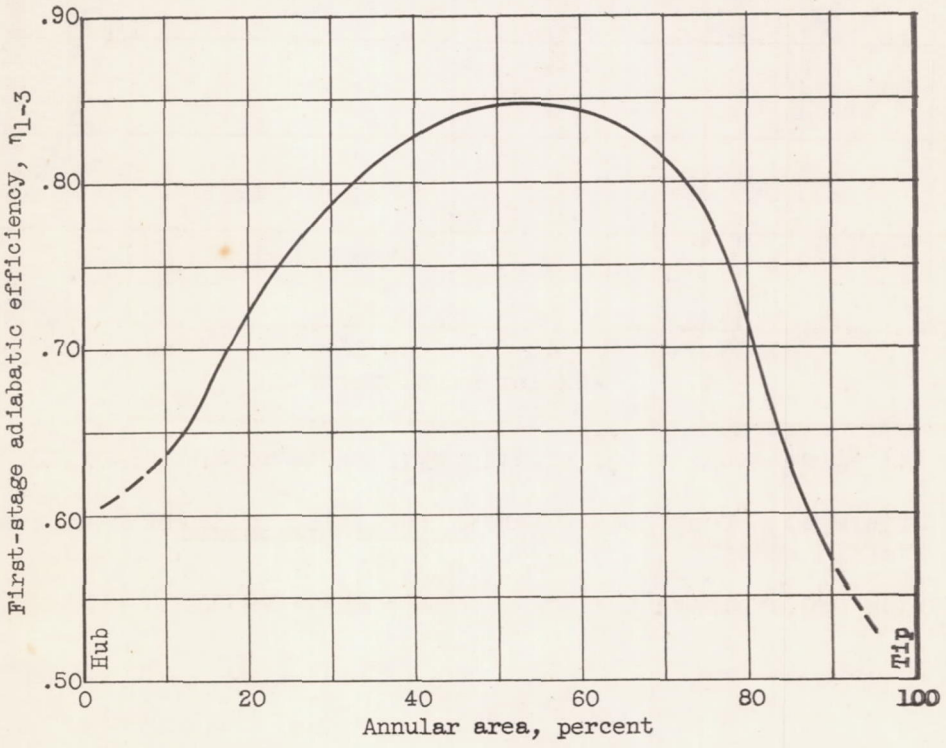
(d) Third-stage work parameter; mass-averaged value, 0.095.

Figure 3. - Concluded. Radial variation of turbine work parameter.

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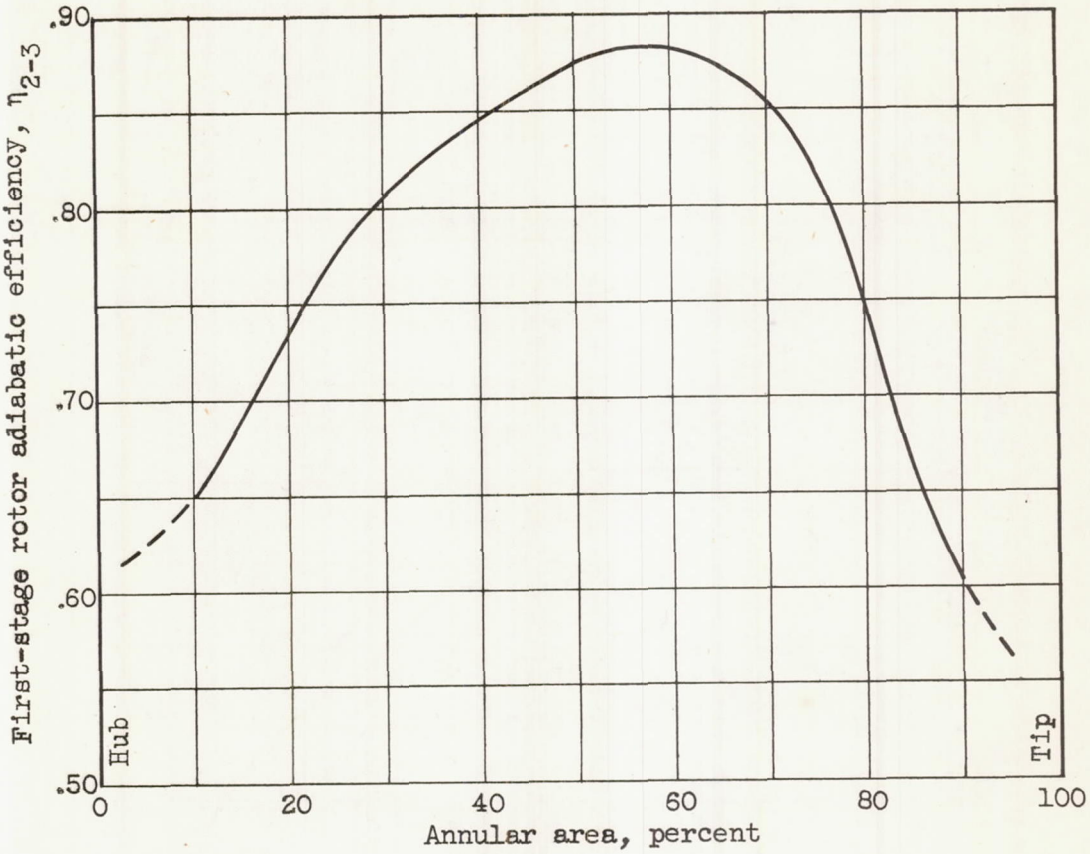
(a) Over-all efficiency; mass-averaged value, 0.83.



(b) First-stage efficiency; mass-averaged value, 0.74.

Figure 4. - Radial variation of turbine adiabatic efficiency.

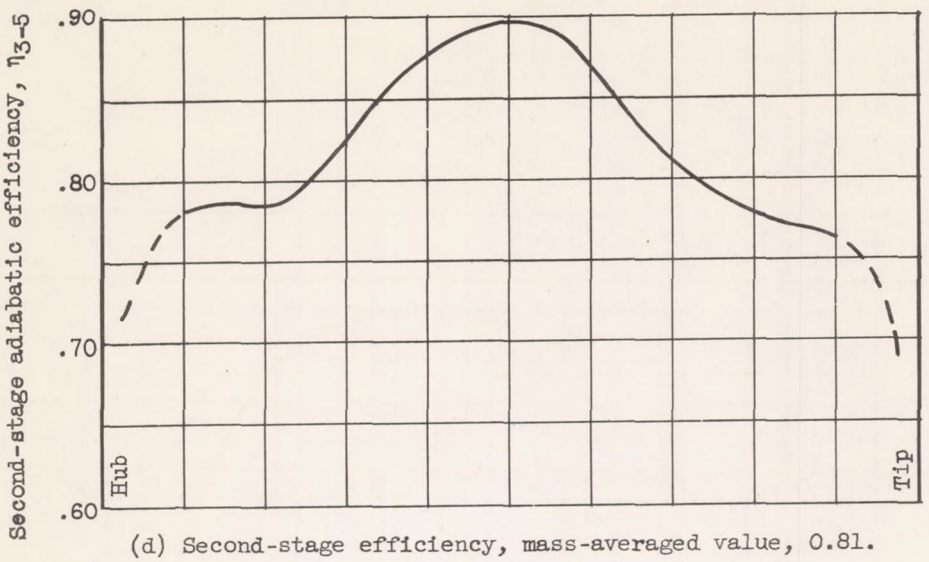
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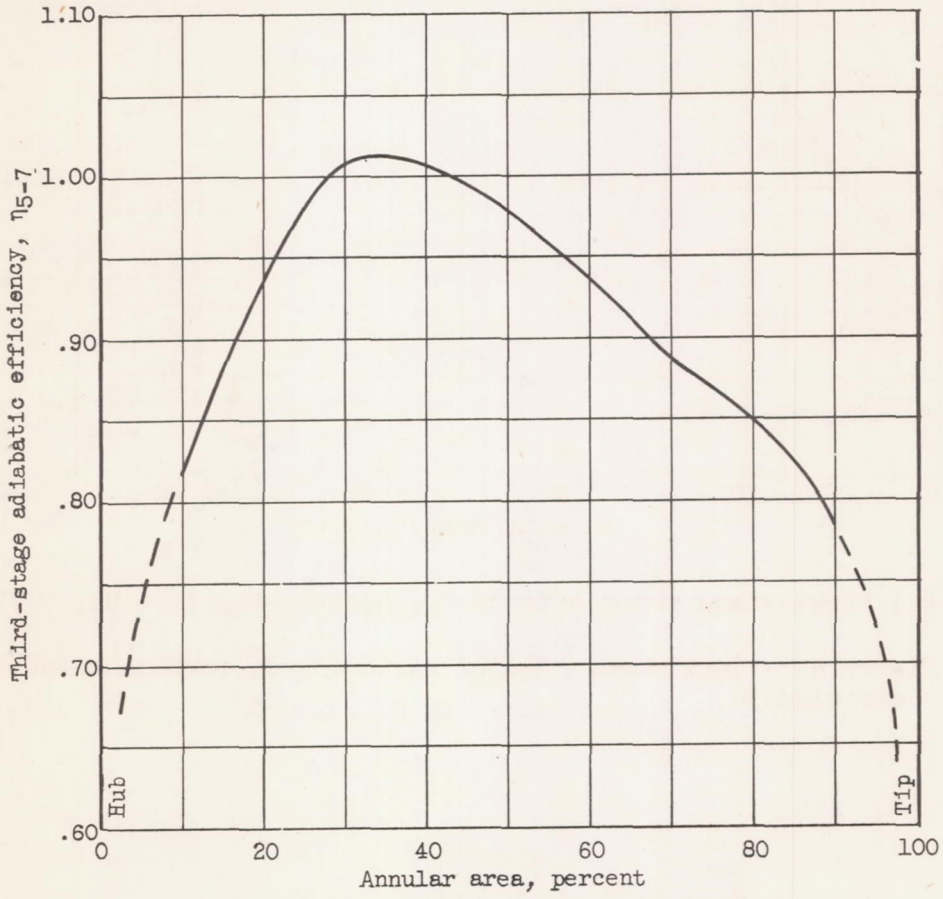
(c) First-stage rotor efficiency; mass-averaged value, 0.77.

Figure 4. - Continued. Radial variation of turbine adiabatic efficiency.

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(d) Second-stage efficiency, mass-averaged value, 0.81.



(e) Third-stage efficiency; mass-averaged value, 0.89.

Figure 4. - Concluded. Radial variation of turbine adiabatic efficiency.

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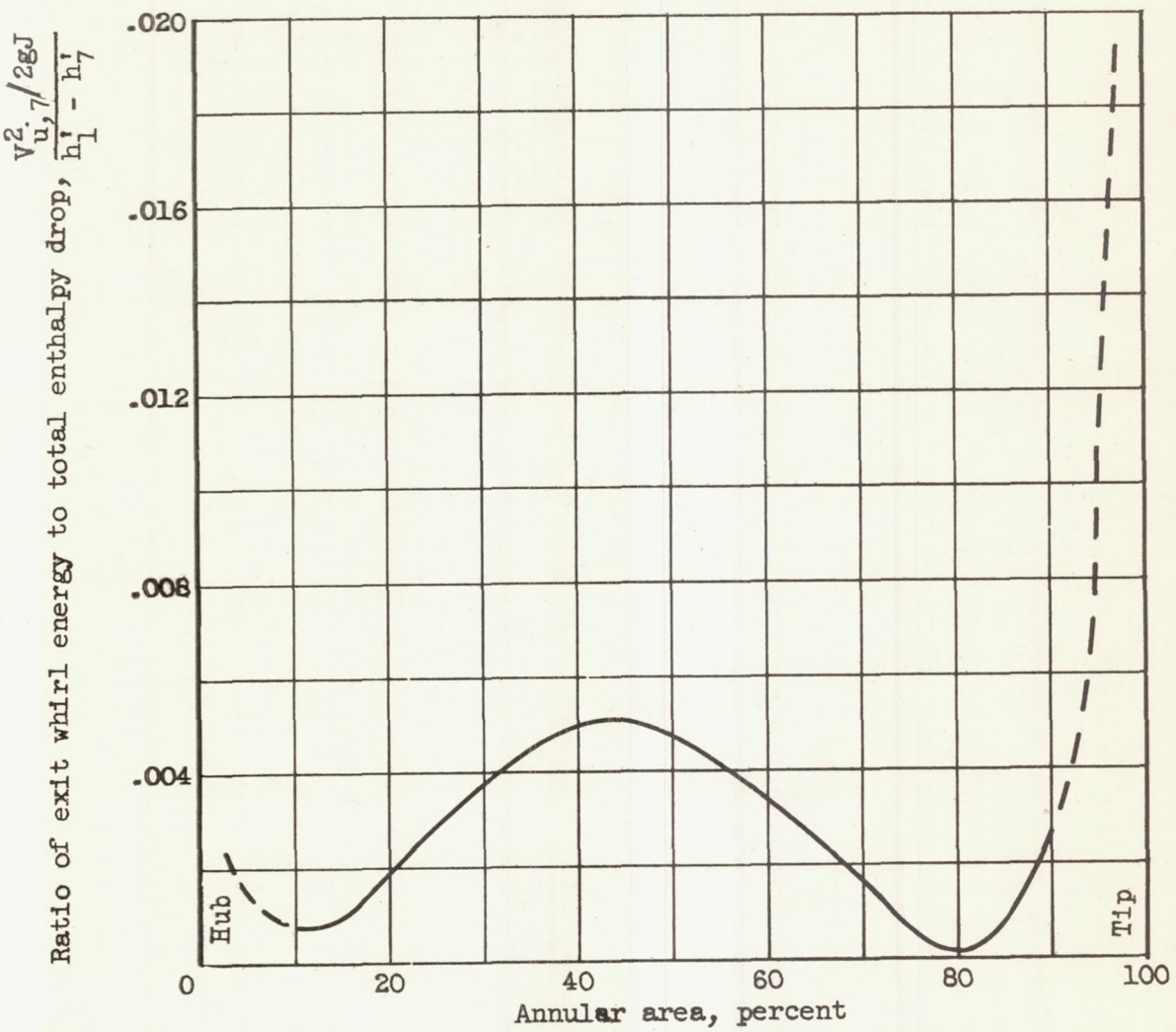


Figure 5. - Radial variation of exit whirl energy loss; mass-averaged value, 0.0034.

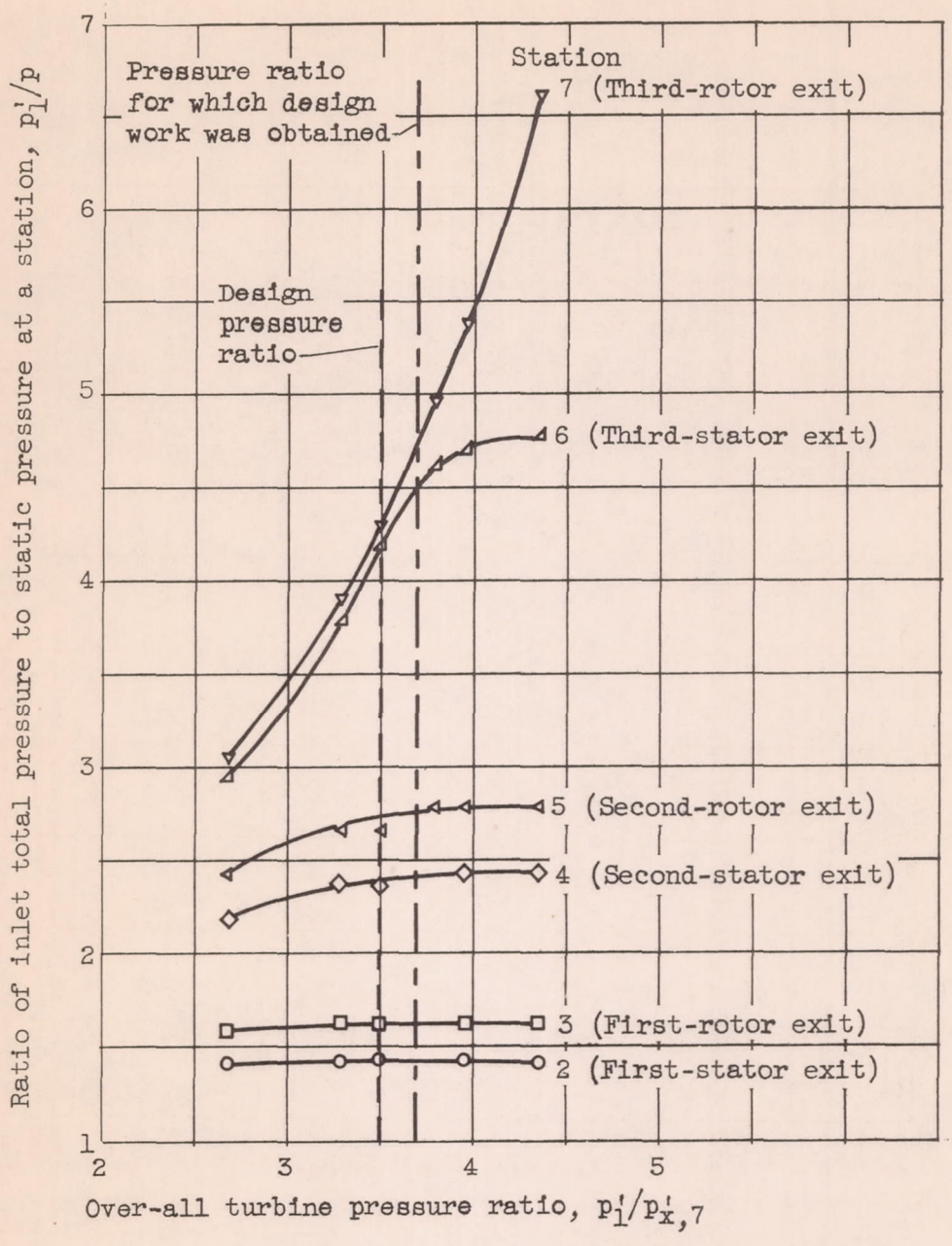


Figure 6. - Variation of ratio of inlet total pressure to static pressure at a station with over-all pressure ratio at design speed.

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